NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

REPORT No. 250

DESCRIPTION OF THE N. A. C. A. UNIVERSAL TEST ENGINE AND SOME TEST RESULTS

By MARSDEN WARE

WASHINGTON
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1927

AERONAUTICAL SYMBOLS

1. FUNDAMENTAL AND DERIVED UNITS

	Symbol	Metric		English	
		Unit	Symbol	Unit	Symbol
Length	l	meter	m	foot (or mile)	ft. (or mi.)
Time	t	second	sec	second (or hour)	sec. (or hr.)
Force	F	weight of one kilogram	kg	weight of one pound	lb.
Power	P	kg/m/sec		horsepower	HP.
Speed		km/hr		mi./hr	M. P. H.
		m/sec		ft./sec	f. p. s.

2. GENERAL SYMBOLS, ETC.

W, Weight, $=mg$

g, Standard acceleration of gravity $=9.80665$ m/sec.$^2=32.1740$ ft./sec.^{2}

m, Mass, $=\frac{W}{g}$

ρ, Density (mass per unit volume).

Standard density of dry air, 0.12497 (kg-m^{-4} sec.^{2}) at 15° C and 760 mm $=0.002378$ (lb.-ft.$^{-4}$ sec.^{2}).

Specific weight of "standard" air, 1.2255 kg/m$^3=0.07651$ lb./ft.^{3}

mk^2, Moment of inertia (indicate axis of the radius of gyration, k, by proper subscript).

S, Area.

S_w, Wing area, etc.

G, Gap.

b, Span.

c, Chord length.

b/c, Aspect ratio.

f, Distance from $c.\ g.$ to elevator hinge.

μ, Coefficient of viscosity.

3. AERODYNAMICAL SYMBOLS

V, True air speed.

q, Dynamic (or impact) pressure $=\frac{1}{2}\rho V^2$

L, Lift, absolute coefficient $C_L=\frac{L}{qS}$

D, Drag, absolute coefficient $C_D=\frac{D}{qS}$

C, Cross-wind force, absolute coefficient $C_C=\frac{C}{qS}$

R, Resultant force. (Note that these coefficients are twice as large as the old coefficients L_C, D_C.)

i_w Angle of setting of wings (relative to thrust line).

i_t, Angle of stabilizer setting with reference to thrust line.

γ, Dihedral angle.

$\rho\frac{Vl}{\mu}$, Reynolds Number, where l is a linear dimension.

e. g., for a model airfoil 3 in. chord, 100 mi./hr. normal pressure, 0° C: 255,000 and at 15° C., 230,000;

or for a model of 10 cm chord 40 m/sec, corresponding numbers are 299,000 and 270,000.

C_p, Center of pressure coefficient (ratio of distance of $C.\ P.$ from leading edge to chord length).

β, Angle of stabilizer setting with reference to lower wing, $=(i_t-i_w)$.

α, Angle of attack.

ϵ, Angle of downwash.

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By MARSDEN WARE
Langley Memorial Aeronautical Laboratory

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

3341 NAVY BUILDING, WASHINGTON, D. C.

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REPORT No. 250

DESCRIPTION OF THE N. A. C. A. UNIVERSAL TEST ENGINE AND SOME TEST RESULTS

By Marsden Ware

SUMMARY

This report describes the 5-inch bore by 7-inch stroke single cylinder test engine used at the Langley Field laboratory of the National Advisory Committee for Aeronautics in laboratory research on internal-combustion engine problems and presents some results of tests made therewith.

The engine is arranged for variation over wide ranges, of the compression ratio and lift and timing of both inlet and exhaust valves while the engine is in operation. Provision is also made for the connection of a number of auxiliaries. These features tend to make the engine universal in character, and especially suited for the study of certain problems involving change in compression ratio, valve timing, and lift.

Incidental to investigations of carburetor and fuel injection engine problems considerable data have been obtained which indicate the effect of changes of compression ratio on friction horsepower and volumetric efficiency. From this and some other work, it appears that with a change in compression ratio from 5 to 13, the friction horsepower obtained by motoring the engine increases by about 15 per cent. The volumetric efficiency of the engine was found to remain practically unchanged between compression ratios of 5.3 and 7.3 with carburetor operation and between 9.5 and 13 with fuel-injection operation.

The results of some tests are presented also that show the power obtained when operating as a carburetor engine on aviation gasoline at compression ratios in excess of that which will permit full throttle as a normal engine and controlling detonation by throttling the intake charge and by varying the inlet valve timing. For fixed compression ratios in these tests throttling gave the least power while variation of the inlet valve closing time with the opening time kept fixed gave the greatest power for the conditions tried.

INTRODUCTION

In order to obtain reliable results from laboratory research, it is necessary to control all variables not under investigation. When dealing with internal combustion engine problems, this is usually very difficult to do. For example, if, in the examination of the effect of a change of compression ratio on engine performance the compression ratio were varied by changing pistons, it would be difficult to obtain the same fit of pistons and rings and to insure identical conditions of lubrication and cooling in the various tests. In addition, atmospheric conditions of pressure, temperature, and humidity would probably change while such alterations were being made and would need to be taken into consideration in the analyses. These difficulties can be eliminated by using an engine specially constructed for changing quickly the variables under investigation without changing other variables.

For the tests mentioned in the previous paragraph, most of the incidental variables would be practically eliminated for direct and succeeding observations if the compression ratio could be altered in a few seconds time and without changing pistons. By the use of suitable auxiliary equipment for the control or the measurement of humidity, pressures, and temperatures of the air and the temperatures of the oil and water, the effects of these variables may be eliminated or determined for protracted series of tests.

Arrangements can be made for the ready alteration of other engine variables and for the connection of various accessories, thus providing a unit suitable for carrying on a large variety of internal-combustion engine research problems with greater facility and with greater reliability of results than would be possible otherwise.

PART I

DESCRIPTION OF THE NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS UNIVERSAL TEST ENGINE

The National Advisory Committee for Aeronautics universal test engine is a single cylinder, four-stroke cycle engine designed especially for laboratory research on internal-combustion engine problems. Figure 1 shows a three-fourths rear view of this engine and Figures 2 and 3 show the general construction. The engine is of rugged design to insure long service without detailed attention. Ball and roller bearings are used in place of plain bearings to a large extent

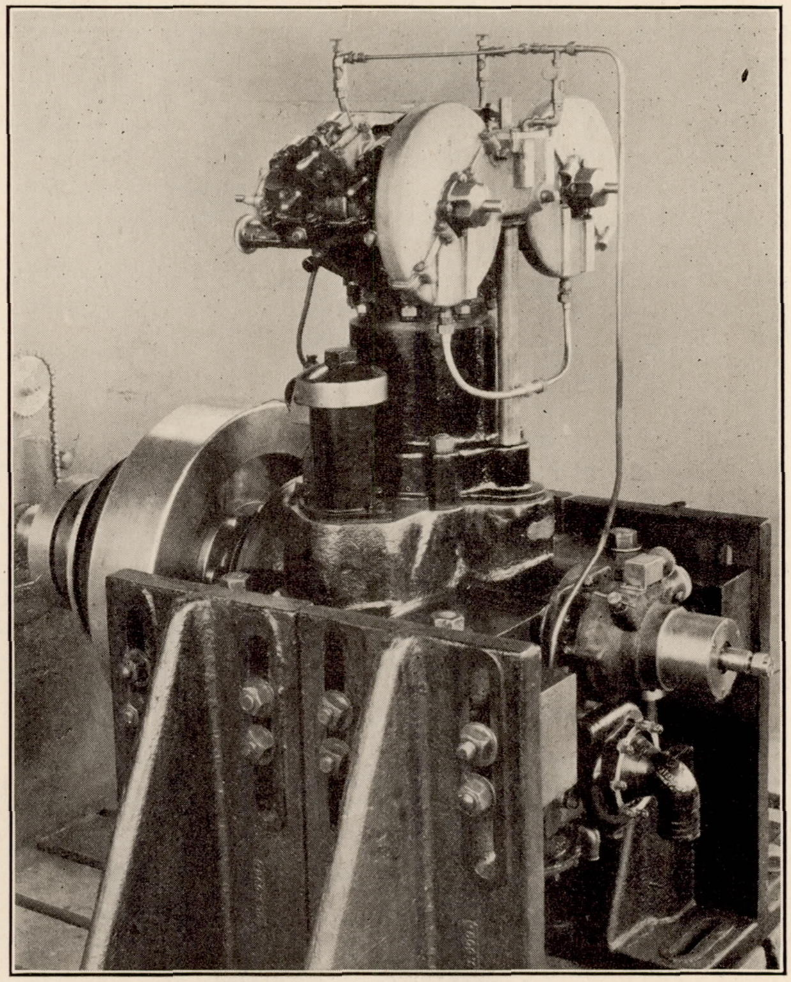

FIG. 1.—N. A. C. A. universal test engine

tending to stabilize and minimize bearing friction losses. The bore of the engine is 5 inches and the stroke is 7 inches. There are two exhaust valves and two inlet valves all of the same size. The valves are operated by rocker arms from overhead camshafts.

Provision is made for altering, during operation, the compression ratio and the opening angle, closing angle, and the lift of both the exhaust and inlet valves. Control of these variables is arranged so that each may be altered independently of the others. The compression ratio may be changed from 4 : 1 to between 13 and 15 : 1 according to the valve lift used and with the piston shown. The opening and closing time adjustments of the inlet and exhaust valves have ranges of 50° each, so that the valve opening periods may be increased a total of 100° from a minimum of about 200°. The timing corresponding to the minimum position may be

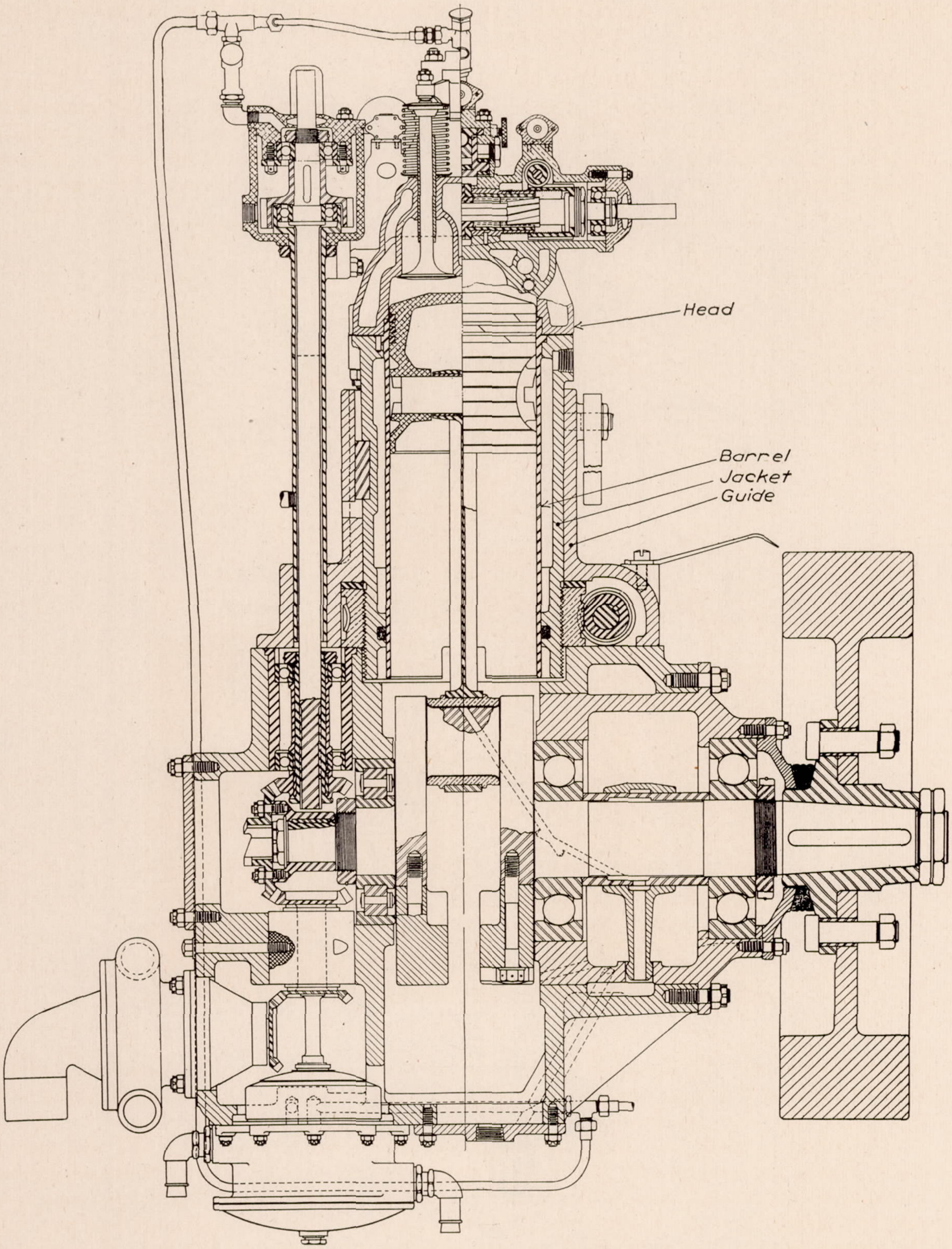

FIG. 2.—N. A. C. A. universal test engine—longitudinal cross section

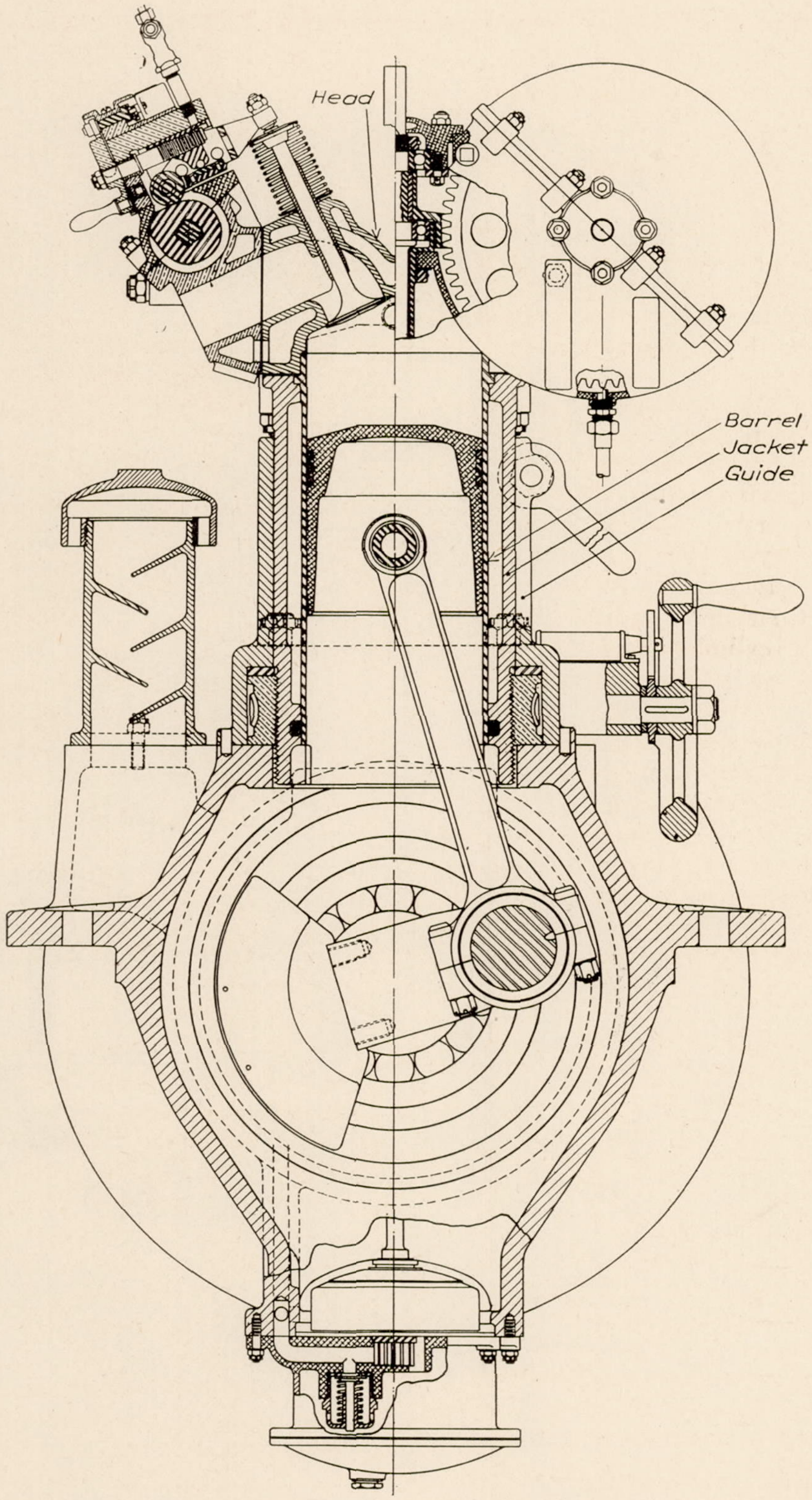

FIG. 3.—N. A. C. A. universal test engine—transverse cross section

altered by changing the setting of the valve operating gear train in the usual manner. The inlet valve lift may be changed from $\frac{7}{32}$ to $\frac{7}{16}$ inch and the exhaust valve lift from $\frac{7}{32}$ to ½ inch.

The engine construction provides for the connection of various accessory apparatus such as ignition timer, fuel injection pumps, pressure indicating instruments, etc.

COMPRESSION RATIO AND VALVE-TIMING CONTROLS

The mechanisms used for changing the compression ratio and the valve timing are probably the most interesting features of the engine and will be described first.

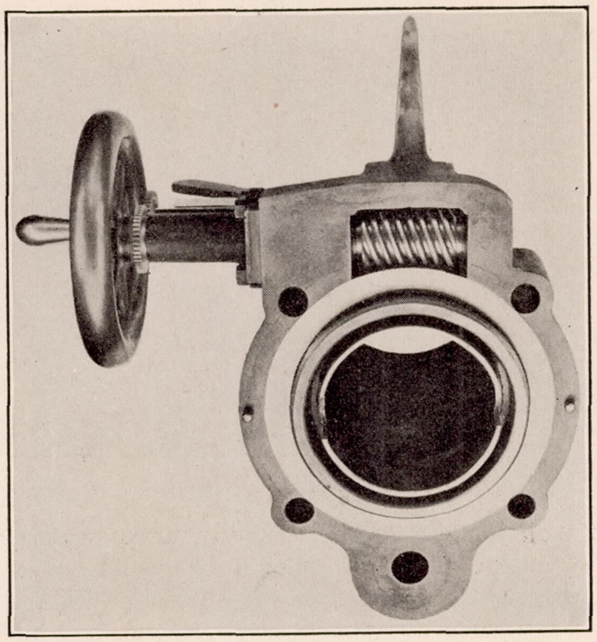

FIG. 4.—View of cylinder guide, jacket, and barrel from bottom, with handwheel and mechanism for effecting change of compression ratio

The compression ratio is varied by moving the cylinder unit vertically, the stroke of the piston remaining fixed, thus changing the clearance volume. The manner in which this is accomplished can be seen by examination of Figures 1, 2, 3, and 4.

There are four main parts to the cylinder construction: The guide, jacket, barrel, and head. The jacket, barrel, and head form an assembled cylinder unit that is movable with respect to the crank case and guide for the purpose of varying the compression ratio. The guide is bolted rigidly to the crank case and provides the means for guiding and fastening the movable cylinder unit. The barrel fits within the jacket, which in turn fits within the guide.

Threads formed on the outside and bottom of the cylinder jacket engage an internally threaded ring, which surrounds the jacket and is restrained from moving vertically by the guide. Rotation of the ring will, therefore, cause the cylinder and head to be raised or lowered according to the direction of rotation, the jacket being prevented from rotating by means of a key fitted permanently in the cylinder jacket, but free to slide in a keyway cut in the cylinder guide. The ring is rotated by a worm meshing with teeth formed on the outside of the ring, a handwheel mounted on the same shaft with the worm providing manual control.

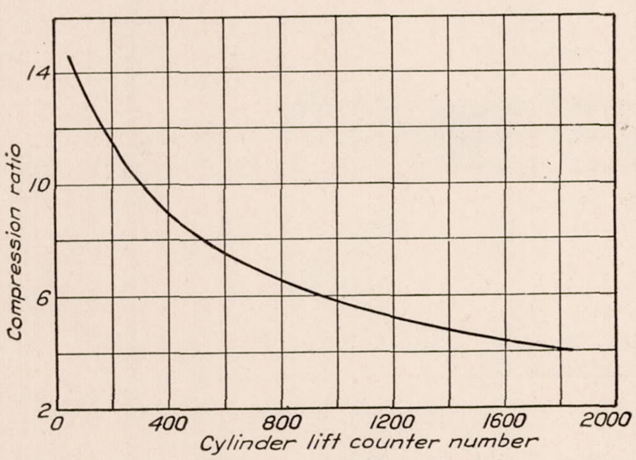

FIG. 5.—Calibration curve of cylinder lift counter

In order to relieve the threaded ring from the necessity of taking the entire explosion load while power tests are being made, the cylinder jacket is clamped in the guide. For this purpose, the cylinder guide is split vertically and is provided with means for clamping the guide around the jacket by the hand lever located on the side of the guide and immediately above the worm and threaded ring construction and shown in Figures 2 and 3.

A counter is geared to the worm shaft to give the position of the cylinder and head with respect to the crank case. The compression ratio is readily ascertained from the counter reading by use of a calibration curve obtained from actual measurement of the clearance volume at a number of different positions of the cylinder. If it is desired to make a test at a definite compression ratio, the cylinder is moved to obtain the counter reading corresponding to this compression ratio. Figure 5 shows a calibration curve for the piston shown in Figures 2 and 3.

The timing of the opening and closing of the valves is varied by the use of three part cams. The central part of each cam is fixed directly to the cam shaft by means of splines while the two outside parts can be rotated independently with respect to the central part and to each other.

All three parts of the cam have the same contour so that when they are aligned they act as a single broad face cam. Displacements of the outside parts from the aligned position are

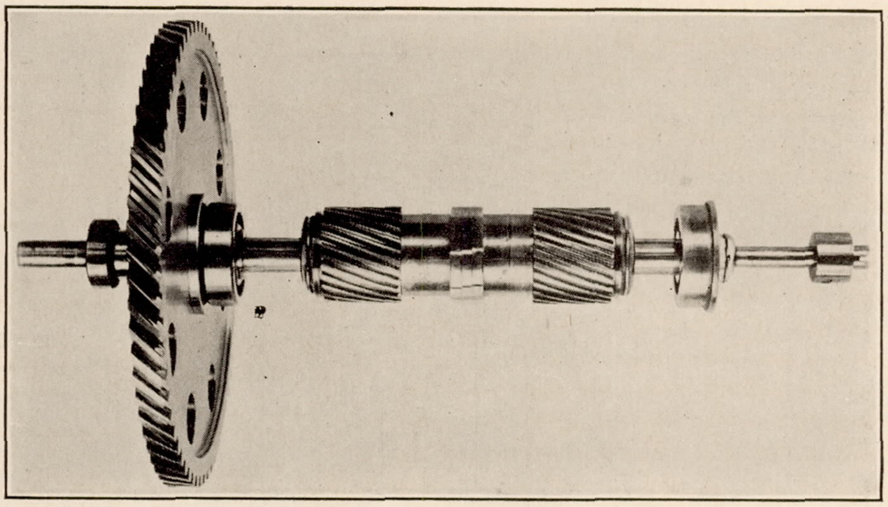

FIG. 6.—Camshaft assembly—cam parts displaced slightly from aligned position

made in opposite directions so that rotation of one of the outside parts varies the opening point while rotation of the other in the opposite direction varies the closing point. When both outside parts are displaced from the aligned position, the dwells of the three parts form one continuous dwell. The dwell of the movable cam parts must be at least equal to the angular variation of the parts from the aligned position. In this engine the dwell of the inlet cam is 50 crank shaft degrees and the dwell of the exhaust cam is 80 crank shaft degrees with both inlet and exhaust cams

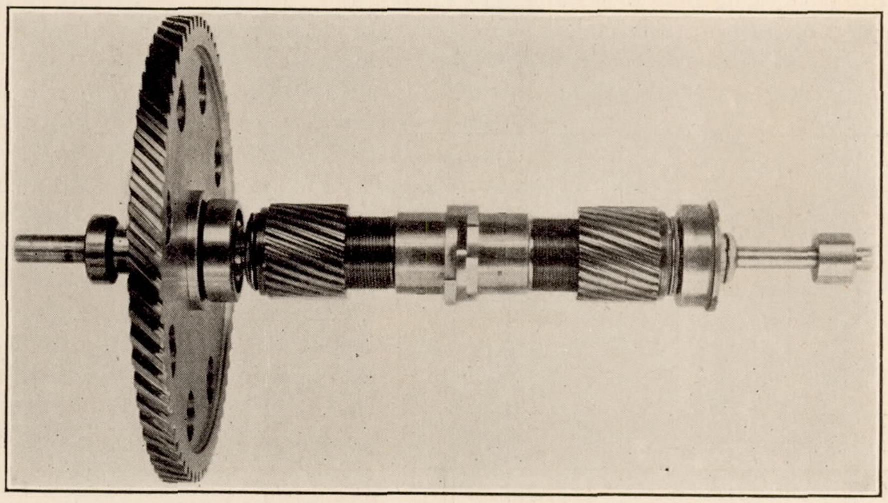

FIG. 7.—Cam-shaft assembly—cam parts displaced their maximum amounts from aligned position

aligned. Figure 6 shows a cam shaft assembly with the three cam parts nearly aligned; Figure 7 shows the three parts at their maximum displacement position.

Figure 8 is a sectional view of the operating mechanism. Each outside part of the cams is formed on the end of a sleeve, A, splined internally with helical splines. A bronze bushing, B, surrounding each sleeve, A, and pinned to the cam shaft housing, C, retains the parts of the cams in their proper axial location on the cam shaft and serves as a bearing for the cam shaft

unit. Another sleeve, D, is located between the cam shaft and the sleeve that carries the outside cam part. This sleeve slides on straight axial splines on the cam shaft, E, while at the same time external helical splines cut on the outside of D engage internal helical splines in A.

As a result of this spline construction, axial movement of sleeve, D, will cause relative rotation between the cam shaft, E, and the sleeve, A. Since the center part of the cam, F, is splined directly to the cam shaft, the outside part of the cam, A, will be rotated relative to the center part, F, and the valve opening or closing point will be changed according to which outside part is rotated.

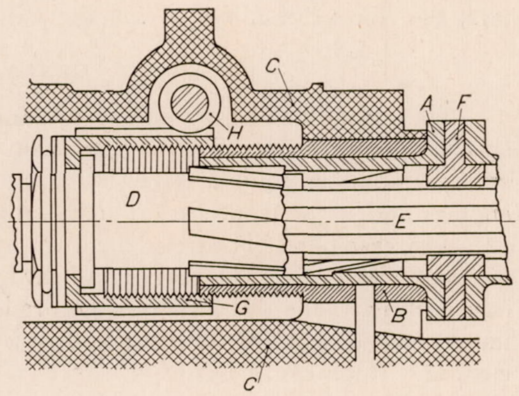

FIG. 8.—Detail of mechanism for making cam adjustment

Sleeve, A, and intermediate sleeve, D, rotate with the cam shaft when the engine is in operation. Since it is desired to vary the valve timing while the engine is in operation, it is necessary to provide for the axial movement of sleeve, D, by some method that will permit its continued rotation during the adjusting operation. A third sleeve, G, provided with an internal flange, serves as a shifting collar for sleeve, D. Internal threads are provided on sleeve, G, which engage with threads formed on stationary sleeve, B, for a considerable portion of its length. Gear teeth cut on the outside of G mesh with a small hand operated worm, H. Rotation of the worm causes G to move axially with respect to the cam shaft in turn shifting D on the cam shaft and causing relative displacement between the parts of the cam whether the cam shaft is turning or is stationary.

The cams operate the valve by direct action on rocker arms, each pair of valves being actuated by a single rocker arm.

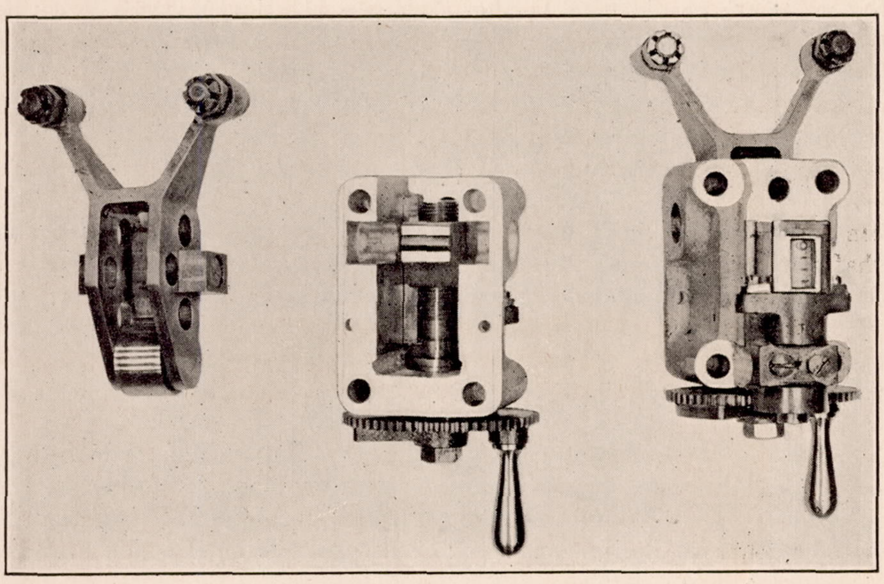
FIG. 9.—Rocker arm and housing

Variation of valve lift is effected by changing the position of the rocker arm fulcrum by means of the mechanism shown in Figures 3 and 9. The fulcrum is formed by a small hardened pin bearing against a hardened plate fastened to the rocker arm, the line of contact between the pin and the plate forming the fulcrum axis. The rocker arm is constrained from changing its position with respect to the valve and cams by trunions machined on the outside of the rocker

arm carried in blocks free to move in the rocker arm housing only and parallel to the valve stem axis. The hardened pin is mounted in a block that is movable with respect to the rocker arm housing. The fulcrum point is changed by turning a small handwheel mounted on a threaded shaft, the threads in turn producing linear movement of the block carrying the fulcrum pin.

All the handwheels controlling the valve functions are geared directly to revolution counters, the counter readings giving the value of the timing and lift of the valves.

GENERAL CONSTRUCTION

The engine has heavy mounting flanges integral with and extending the full length of the cast-iron crank case. This construction permits of flexibility in the mounting of the engine on dynamometer equipment with the use of the usual parts of this equipment for the testing of engines and without alteration of the crank case or provision of a special base.

The crankshaft is carried by two large ball bearings and one roller bearing. The roller bearing is located on the gear end of the crank shaft and is mounted directly in the main crank case casting. The two ball bearings are located on the fly wheel end of the crank shaft and are mounted in a cylindrical cast-iron cage which is mounted in and bolted to the main crank case casting. Removal of this cage effects removal of the crank shaft and bearing assembly from the crank case.

The cylinder structure has been described partially under the description of the compression ratio changing mechanism. A long annular space is formed between the jacket and the barrel which provides for circulation of the cooling water. The cylinder head has a cored water jacket with no internal communication with the cylinder jacket. This construction permits separate control of the flow of the cooling water to the cylinder and head and consequently permits maintaining different water temperatures for the two parts. The guide, jacket, and head are iron castings and the barrel is a steel forging.

Three openings are provided in the head for the insertion of spark plugs, fuel injection valves, pressure indicating devices, etc. These openings are provided with standard metric spark plug threads. Two openings are located on opposite sides of the head; the third opening is located in the center of the head. Several threaded openings into the jacket and head are provided for connecting cooling water fittings.

The piston and connecting rod shown in Figures 2 and 3 are standard Liberty engine parts.

Two cam shafts are provided, one for the inlet valves, the other for the exhaust valves. These two shafts are driven from a vertical shaft by helical gears. The vertical shaft is in turn driven by the crank shaft through bevel gears. The vertical shaft bevel gear is mounted directly on a relatively short hollow shaft having internal splines which engage splines on the lower end of the vertical shaft proper. This construction permits the upper shaft to move vertically with the cylinder. The crank shaft bevel gear furnishes also the drive means for the water and oil pumps.

The ends of both cam shafts and the top end of the cam driving vertical shaft extend outside their housings to provide driving means for various auxiliaries, such as ignition timing mechanisms, indicators, etc. The crank shaft bevel gear is splined internally to provide a driving means for other auxiliaries, such as fuel injection pumps, which may be fastened to the rear of the engine base.

The oil pump is a combined scavenging and pressure pump. The scavenged oil is carried off to an external reservoir. The oil from the pressure side of the pump is carried through drilled passages in the base to the front of the engine. The path divides at this point, one path leading to the cam shaft mechanism, where the flow to the various parts is regulated by three sight-feed adjustable oilers, the other path leading to the crank shaft through a bronze casting that rides on the crank shaft. Passages are drilled in the crank shaft for the lubrication of the connecting rod bearing. The bearing in the crank case, the piston pin, and piston are lubricated by oil thrown off the connecting rod bearing.

ENGINE TEST EQUIPMENT

In the use of this engine for research, certain auxiliary apparatus are necessary for measuring various quantities affecting the performance, or entering the performance determinations. While the number and type of such auxiliaries will vary with the kind of work, or the particular investigation undertaken, there are a number that enter into a large percentage of the work and may consequently be considered to be combined with the engine to form a research unit. The chief auxiliary items of equipment used with this engine are an electric dynamometer, a gasometer, fuel and water measuring apparatus, air, oil, and water temperature regulating apparatus, exhaust handling equipment, and special carburetors and fuel injection apparatus.

The electric dynamometer not only serves as a readily adjustable and convenient brake for absorbing the power developed by the engine, but also provides the means for motoring the engine. The torque required to motor the engine or that produced by the engine when it is developing power, appears as torque on the dynamometer field housing. The housing is mounted on ball bearings and therefore can swing free in its support pedestals. An arm fastened to the housing acts on a scale which resists and indicates the torque on the housing. The power developed in power tests or that required in motoring tests is determined from the torque and speed of revolution. The speed of revolution is determined by a tachometer and an electrically operated revolution counter and stop watch.

The gasometer or displacement meter is used in connection with an automatic timing mechanism for giving the time for the consumption of a given volume of air.

Three methods are used in measuring fuel-consumption rates. One method consists of the measurement of the time for the consumption of a given volume of fuel. This method requires frequent determinations of specific gravity of fuel used where weight of fuel is desired. The second method consists of the measurement of the time for the consumption of a given weight of fuel, measures the weight directly, and is used for most work. A third method employs a calibrated orifice where the rate of flow is a function of the loss of head in the passage of the fuel through the orifice. This method is valuable as a means of indicating quickly the approximate rate of fuel flow following carburetor adjustment and is used in conjunction with one of the other more accurate methods.

An electric resistance type air heater is provided between the gasometer and the carburetor to enable the inlet air temperature to be controlled. Oil and water temperature regulating means are provided also for controlling these temperatures. Temperature indicating devices are provided at appropriate locations. A blower and piping system are used to carry the exhaust gases away from the engine and deliver them outside of the building.

In the comparison of the relative performance with two different fuels, a carburetor has been used that has two throttles and two distinct metering systems and float chambers, but a single air inlet and single mixture exit. One fuel, which may form a standard of comparison, is used in one side of the carburetor, while another fuel, which may be the fuel to be compared, is used in the other. The controls of both throttles are carried to the control board, and the operator may change fuels instantaneously by opening one throttle and closing the other.

Fuel injection pumps are attached to the engine for fuel injection engine work. In this work the fuel fed to the pump usually is required to have a pressure of over 100 lb./sq. in. A small gear pump used for this purpose, fuel weighing apparatus, and supply tank are mounted on a stand, forming a compact primary fuel system.

PART II

SOME RESULTS OF TESTS WITH UNIVERSAL TEST ENGINE

An engine of this kind is especially suited for the study of a number of problems that have received some attention in the past, but under conditions which in most cases have, of necessity, included undesirable variables. Of interest among such problems is the effect of changes in compression ratio on volumetric efficiency and engine friction. While this engine has been used very little in the direct study of these problems, considerable data having a bearing on these relationships have been obtained incidental to work on other problems and have been assembled for presentation in this report. Some results obtained in an examination of the ground level operation of the over-compressed engine in which the variable valve timing features entered are also presented.

VOLUMETRIC EFFICIENCY AND COMPRESSION RATIO

Air measurements taken during a large number of tests made in connection with a carburetor engine problem involving a total of 83 observations at compression ratios of 5.3, 6.3, and 7.3 are used in the consideration of the effect of compression ratio changes upon volumetric efficiency. Volumetric efficiency as used in this report is defined as the ratio of the actual volume of air drawn in by the engine on each cycle when reduced to the temperature and pressure conditions of the air at the engine to the displacement volume of the engine. No difference in volumetric efficiency in excess of 1 per cent was noted between the average values for the three compression ratios.

Air measurements taken in connection with a fuel injection compression ignition problem, involving 80 observations and covering compression ratios from 9.5 to 13, show a similar effect.

The results of the two series of tests are not directly comparable since any influence on volumetric efficiency of fuel vaporization on the suction stroke in the carburetor work does not appear in the fuel injection work with the fuel injected near top dead center on the compression stroke. Consequently no attempt is made to estimate the effect over the entire range of compression ratios from 5 to 13. The test conditions in each set were maintained practically constant with sufficient variation in air-fuel ratio in the carburetor runs to permit elimination of the effect of this variable in this analysis. The two sets of results are, therefore, comparable in themselves.

Ricardo gives results of a test of the effect of a change in compression ratio from 5:1 to 7:1 in which the rate of air flow was reduced from 209.5 lb./hr. to 190 lb./hr.—a change of 10 per cent. (Reference 1.) No difference of this magnitude was noted for any of the large number of points providing the information presented in the previous paragraph and it appears evident that in so far as these tests on the universal test engine are concerned the effect of compression ratio on volumetric efficiency, for a considerable change in this ratio, is so small that it is of little practical importance.

ENGINE FRICTION AND COMPRESSION RATIO

A large number of tests have been made in both carburetor and fuel-injection work during which measurements of friction horsepower were made by motoring the engine with the dynamometer. These tests covered compression ratios from 5.3 to 7.3 and 9.5 to 13. The results of these friction measurements plotted against compression ratio, together with a series of independent motoring tests for the complete range of compression ratio from 5 to 13, are given in Figure 10. The independent motoring tests were made with oil and water temperatures maintained by means external to the engine. Four distinct groups of results are given for the same speed of revolution but with oil and water temperature conditions different between the groups although constant for each group.

Curves A and B give friction mean effective pressures obtained from the friction horsepower tests taken immediately after power tests, while curves C and D apply to the independent motoring tests. Curve A is the average curve for the result obtained in connection with full throttle power runs on a carburetor engine problem and was obtained by careful consideration of over 104 test points. Curve B is the average curve for the results obtained in connection with a fuel-injection engine problem and includes 78 test points. Curve C is the average curve of a series of independent motoring tests during which the compression ratio range shown was traversed repeatedly in opposite directions and the observations taken as rapidly as possible. None of the points taken departed over 2 per cent from the curve shown. The two points designated D are data for independent motoring tests, taken some time previous to the other group.

While there is considerable vertical displacement between the groups, they are consistent as regards slope and it is evident that friction horsepower, as measured by motoring tests, increases with increase in compression ratio. The vertical displacement between curves A and B is probably due to a difference in kind and temperature of lubricating oil used in the two groups of tests and to fits of pistons and rings which were changed during the time that elapsed between the two groups of tests. Both curve A and curve B show practically the same rate of

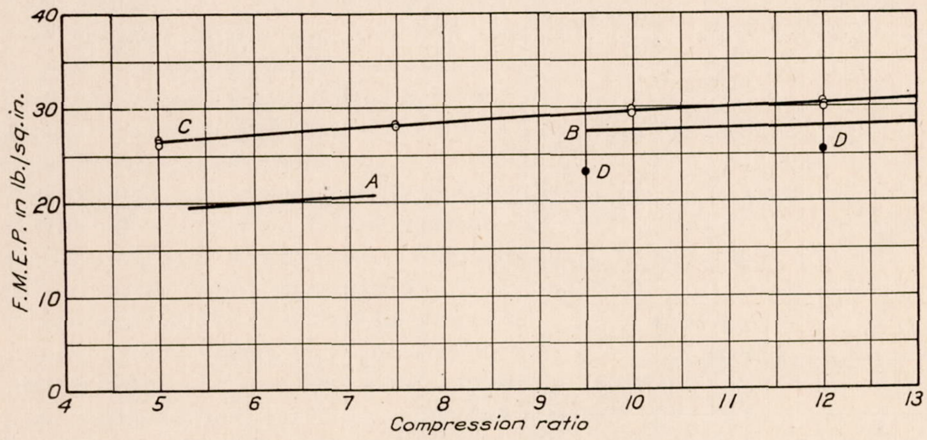

FIG. 10.—Effect of change in compression ratio upon F. M. E. P.

increase in friction as curve C over the range of compression ratio covered by A and B. As a result of this, it is assumed that curve C is a fair representation of the order of change in friction with change in compression ratio for observation taken immediately after power runs. This curve shows an increase in friction of 15 per cent for a change in compression ratio from 5 to 13.

The extent to which these friction results can be used to give the difference between indicated and brake power for actual power operation can not be stated definitely from theoretical considerations alone. In motoring tests, the increase in friction with increase in compression ratio is due primarily to two causes. First, the increased pressures accompanying increased compression ratio will cause greater mechanical friction losses, due primarily to greater piston ring friction and piston side thrust. Second, the operations of suction, compression, expansion, and exhaust during the motoring tests may be such as to require greater work input with increased compression ratio. Due to leakage past the piston and rejection of heat to the jacket, all of the work of the compression stroke will not be returned on the expansion stroke. The difference between the work for these two strokes will increase with increase in compression ratio.

When the engine is under power the conditions are somewhat different. Piston side thrust will increase more with a change in compression ratio when the engine is developing power than in motoring tests where an increase in compression ratio is accompanied by an increase in mean effective pressure. Indicator cards taken under power show the work done by the engine on the compression and expansion strokes and that required to overcome pumping losses on the exhaust and inlet strokes. While there are actual losses due to heat rejection and

leakage past the piston during the compression and expansion strokes in power runs, indicator cards taken during power runs do not show the work required as a result of these losses as they do in motoring tests. Friction power taken from differences of indicator measurements, and brake measurements gives then the purely mechanical losses while the friction power taken from motoring tests involves a lower mechanical loss, in so far as the lesser loads produce lower friction, and also includes the lost work that results from piston leakage on the compression and expansion strokes. Brief theoretical consideration of the influence of change in compression ratio on the components of friction discussed indicates that the order of the change of friction with change in compression ratio from indicator and brake measurements may be the same as that obtained from motoring measurements.

The relation between the friction losses of an engine as determined by motoring test and its friction when developing power can be determined by tests in which accurate indicator diagrams are taken. Tests have been made by the Air Ministry of Great Britain (Reference 2) for the purpose of investigating the relation between friction power losses as measured by motoring tests and those obtained by taking differences between indicator and brake power measurements. These Air Ministry tests were made at only one compression ratio and show the effect of the character of the combustion upon friction determinations by the two methods. At high loads and with a retarded spark the two methods gave similar results; with normal spark the motoring method gave results 15 to 20 per cent lower than the indicator method, and for advanced spark the difference was increased to 20 to 45 per cent.

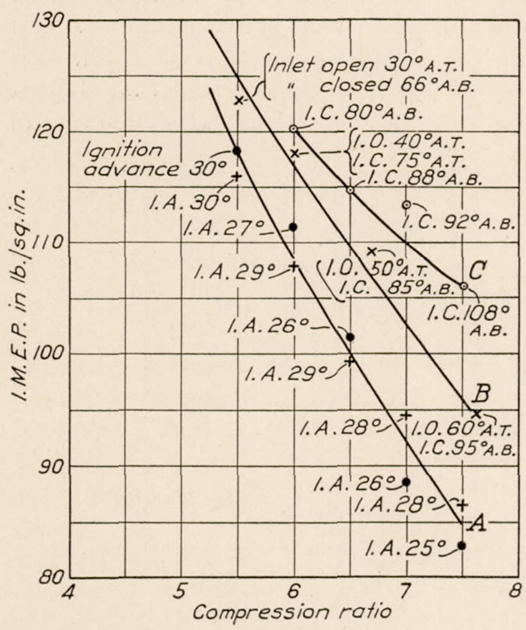

FIG. 11.—Effect of change in compression ratio upon I. M. E. P. when audible detonation is limited by closing throttle or changing inlet valve timing

Motoring power is used as friction horsepower in nearly all test work so that the results obtained with this universal engine and presented herewith are of interest and may apply for certain kinds of combustion. The order of the change in friction for the range in compression ratio investigated in these tests is such that future considerations of engines having compression ratios much higher than used in present engines would make it desirable to make comparisons at a number of compression ratios like those made at a single compression ratio by the Air Ministry.

INLET VALVE TIMING AND THE HIGH COMPRESSION ENGINE

Figure 11 gives indicated mean effective pressures that were obtained at various compression ratios using aviation gasoline for fuel and limiting the detonation at the higher ratios to that considered permissible for continuous operation at a low ratio. These results are presented here as an illustration of the manner in which the variable valve timing mechansim proves valuable in engine research.

Results were obtained for three different methods of limiting detonation: Throttling (curve A), retarding the opening and closing times of the inlet valve (curve B), retarding the closing time only of the inlet valve (curve C). With each method the control was set so that the spark could be advanced beyond the point where maximum power was obtained without causing a perceptible increase in detonation.

These three methods provide different means for controlling the overcompressed engine so that it will not detonate at sea level when using aviation gasoline. When using the first method, the throttle would be opened as the altitude of operation is increased. With the second method, the whole inlet valve timing would be advanced as the altitude is increased.

With the third method only the inlet valve closing time would be advanced as the altitude is increased. The first method has been used considerably, but from these tests it is seen to give the least power at ground level. Figure 11 shows that the second method gives better results than the first. The third method gives the greatest power at ground level, but involves greatest complication of control. Regardless of which method is employed, the same power can be obtained at and above that altitude at which the throttle may be opened wide for the case where it is used to limit detonation and at which the valve timing will be the same as in the throttling case, for the two cases where valve timing is used to limit detonation. A method of limiting detonation by changing the valve timing has been used in the Bristol Jupiter engine and good results have been obtained.

Both the second and third methods are the same in effect in that the delay in closing the inlet valve results in some of the charge drawn in the cylinder being returned to the inlet manifold and the effective compression stroke reduced. Engines so constructed would have, in effect, a lower compression ratio than expansion ratio at low altitudes. The first method involves a similar reduction in volumetric efficiency, but since the compression ratio has not been changed the ratio of the temperatures at the beginning and end of the compression stroke and the compression temperature will be higher, requiring somewhat greater reduction in volumetric efficiency to limit the detonation to the same extent. The ratios of residual gases retained in the cylinder to fresh charge are probably different in the three cases and the difference in power noted is due to this condition to some extent. The first method also involves higher pumping losses than the third.

CONCLUSIONS

The test results presented in this report serve as examples of the use of this universal engine in some research problems where the variable features are of especial service. Examination of the data used in determining the effect of compression ratio on volumetric efficiency together with consideration of conflicting results obtained elsewhere indicate that no general conclusion should be drawn as to the effect of compression ratio on volumetric efficiency. It is evident, however, that an increase in compression ratio from 5 to 13 causes an appreciable increase in friction horsepower as measured by motoring tests.

This engine lends itself to the direct study of such variables. Further study of friction losses from indicator diagrams would determine the relation between the actual losses and those obtained by motoring tests for definite conditions.

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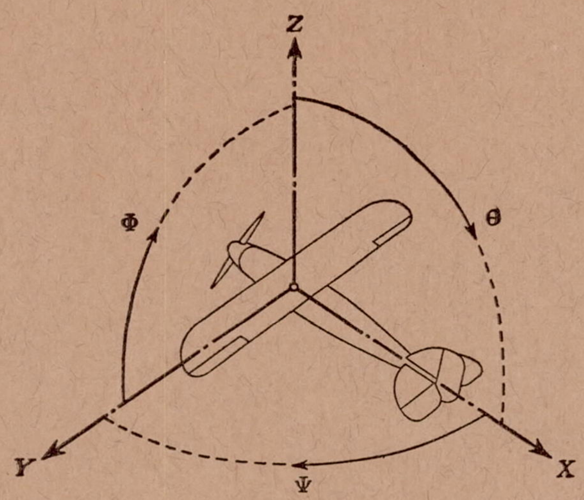

Positive directions of axes and angles (forces and moments) are shown by arrows

Axis		Force (parallel to axis) symbol	Moment about axis			Angle		Velocities	
Designation	Symbol		Designation	Symbol	Positive direction	Designation	Symbol	Linear (component along axis)	Angular
Longitudinal	X	X	rolling	L	$Y \longrightarrow Z$	roll	Φ	u	p
Lateral	Y	Y	pitching	M	$Z \longrightarrow X$	pitch	Θ	v	q
Normal	Z	Z	yawing	N	$X \longrightarrow Y$	yaw	Ψ	w	r

Absolute coefficients of moment

$$C_L=\frac{L}{qbS}\quad C_M=\frac{M}{qcS}\quad C_N=\frac{N}{qfS}$$

Angle of set of control surface (relative to neutral position), δ. (Indicate surface by proper subscript.)

4. PROPELLER SYMBOLS

D, Diameter.
p_e, Effective pitch
p_g, Mean geometric pitch.
p_s, Standard pitch.
p_v, Zero thrust.
p_a, Zero torque.
p/D, Pitch ratio.
V', Inflow velocity.
V_s, Slip stream velocity.

T, Thrust.
Q, Torque.
P, Power.
(If "coefficients" are introduced all units used must be consistent.)
η, Efficiency $= T\,V/P$.
n, Revolutions per sec., r. p. s.
N, Revolutions per minute., R. P. M.
Φ, Effective helix angle $=\tan^{-1}\left(\frac{V}{2\pi rn}\right)$

5. NUMERICAL RELATIONS

1 HP = 76.04 kg/m/sec. = 550 lb./ft./sec.
1 kg/m/sec. = 0.01315 HP.
1 mi./hr. = 0.44704 m/sec.
1 m/sec. = 2.23693 mi./hr.

1 lb. = 0.4535924277 kg.
1 kg = 2.2046224 lb.
1 mi. = 1609.35 m = 5280 ft.
1 m = 3.2808333 ft.